NATIONAL BUREAU OF STANDARDS REPORT

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PROGRESS REPORT

Air Conditioning in Underground Structures

November 1, 1952 to April 30, 1953.

by

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NBS

U. S. DEPARTMENT OF COMMERCE
NATIONAL BUREAU OF STANDARDS

PROGRESS REPORT

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I. INTRODUCTION

During the period from November 1, 1952 to April 31, 1953, tests have been continued in the underground test chamber at Mount Weather, mathematical approaches to the theory of heat transfer to a rock mass bounding an underground space have been studied and are compiled and presented in this report, data from previous tests were analyzed, revisions were made in the Engineering Manual, Part XVI, Chapter 3, and preparations were made for future testing at Mount Weather, Va. and Fort Ritchie, Maryland.

II. TESTS PERFORMED IN UNDERGROUND TEST CHAMBER

Test Condition 8 - Determination of heat and moisture load without ventilation or simulated occupancy from November 10 to November 26 while maintaining constant conditions of 75° DB and 50% R.H. with the air conditioning system.

Test Condition 9 - Conditions were the same as test condition 8 except condenser water reheat was used to reduce the electric reheat November 26 to December 5.

Test Condition 10 - Steady state heat at 75°F air temperature from December 5 to January 5, 1953. No dehumidification.

Test Condition 11 - Steady state heating at 75°F air temperature with ventilation air from January 5 to February 11.

Test Condition 12 - Test condition 10 repeat from February 11 to March 30.

Test Condition 13 - Test condition 11 repeat now in progress.

III. MATHEMATICAL APPROACHES TO HEAT TRANSFER TO UNDERGROUND SPACES

A review of the mathematical approaches to the transfer of heat to the rock surrounding an underground chamber shows that mathematical treatment of the elementary shapes such as the plane surface, circular cylinder and sphere bounded by a medium approaching infinite thickness is found in Carslaw and Jaeger, "Conduction of Heat in Solids". While for the most part the actual equations involving heat transfer are complicated, they may be reduced by numerical integration or approximation and plotted in graphical or tabular form for design application.

Configuration of actual underground installations may be approximated by one or more of the elementary shapes. Considering an underground room to be an assembly of plane surfaces (floor, ceiling and walls) neglects the heat flow into the corners and edges, whereas likening it to a cylinder (lateral surface) neglects the heat flow into ends of the cylinder. A sphere would rarely be an approximation for an actual installation.

To the above named shapes two different boundary conditions can be applied; namely, the constant heat flux case and the constant surface temperature case.

I. Heat transferred to a solid bounding an elementary shape with a constant heat flux and the initial temperature of the solid equal to zero on an arbitrary datum plane.

A. Linear heat flow to a medium of semi-infinite depth from a plane surface.

1. Temperature at depth x

$$\theta = \frac{Q}{k} \sqrt{\alpha t} \left(2 \text{ ierfc } \frac{x}{2\sqrt{\alpha t}} \right)$$

Values of '2 ierfc' of the argument are shown in tabular form in Table 2.

2. Temperature at x = 0.

$$\Theta = \frac{2Q}{k}\sqrt{\frac{\alpha t}{\pi}} = 1.1284 \; \frac{Q}{k}\sqrt{\alpha t}$$

B. Heat flow to the solid bounded internally by a circular cylinder (radius = a).

1. Temperature at $r > a$ in the operational form

$$\bar{\Theta} = \frac{Q}{k}\,\frac{K_0(qr)}{pq\,K_1(qa)}\,, \qquad q^2 = p/\alpha$$

Solution of this operational form can be made for small values of time by use of asymptotic expansions of the Bessel functions and the use of a Laplace transform

2. Temperature at r = a.

$$\bar{\Theta} = \frac{Q}{k}\;\frac{K_0(qa)}{pq\,K_1(qa)}$$

Solution for small values of time:

$$\Theta = \frac{aQ}{k}\left(\frac{2\sqrt{B}}{\sqrt{\pi}} - \frac{B}{2} + \frac{B^{3/2}}{2\pi^{1/2}} - \frac{3}{16}B^2\right)$$

where $B = \dfrac{\alpha t}{a^2}$ and is restricted to values less than 0.3. Values of the function of B in the parenthesis appear in Table 1.

C. Heat flow to the solid bounded internally by a sphere (radius = a).

1. Temperature at radius $r > a$.

$$\Theta = \frac{Qa}{k}\left\{ \operatorname{erfc}\left(\frac{r-a}{2\sqrt{\alpha t}}\right) - e^{\left(\frac{r-a}{a} + \frac{\alpha t}{a^2}\right)} \operatorname{erfc}\left(\frac{r-a}{2\sqrt{\alpha t}} + \sqrt{\frac{\alpha t}{a}}\right)\right\}$$

Values of 'erfc' of the argument are shown in tabular form in Table 2.

2. Temperature at radius r = a.

$$\Theta = \frac{Qa}{k}\left[1 - e^{\frac{\alpha t}{a^2}} \operatorname{erfc} \left(\frac{\alpha t}{a^2}\right)^{1/2}\right]$$

II. Heat transferred to a solid bounding an elementary shape with a constant surface temperature and the initial temperature of the solid equal to zero on an arbitrary datum plane. It must be noted that for these conditions the heat flux is infinite for time equal to zero and therefore the equations are valid for t greater than zero.

A. Linear heat flow to a medium of semi-infinite depth from a flat plane.

1. Temperature at depth x.

$$\Theta = V \operatorname{erfc} \frac{x}{2\sqrt{\alpha t}}$$

2. Heat flux at x = 0

$$H = \frac{kV}{\sqrt{\pi \alpha t}}$$

B. Heat flow to a solid bounded internally by a circular cylinder (radius = a)

1. Temperature at radius $r > a$.

$$\Theta = V\left(\frac{a}{r}\right)^{1/2} \operatorname{erfc}\left(\frac{r-a}{2\sqrt{\alpha t}}\right) + \frac{V(r-a)}{8\sqrt{\frac{ar^3}{\alpha t}}}\left(2 \operatorname{ierfc} \frac{r-a}{2\sqrt{\alpha t}}\right.$$

reasonable for $t < 20{,}000$ hrs., and $a > 10$ ft.

2. Heat flux at r = a

$$H = \frac{4kV}{a\pi^2} \int_0^\infty \frac{e^{-\alpha u^2 t}}{J_0^2(ua)+Y_0^2(ua)} \frac{du}{u}$$

$$= \frac{4kV}{a^2} \; I \left(0,\ 1;\ \frac{\alpha t}{a^2}\right)$$

where values of the integral

$I\left(0,\ 1;\ \frac{\alpha t}{a^2}\right)$ appear in tabular form

in Table 2.

C. Heat flow to a solid bounded internally by a sphere (radius = a).

1. Temperature at radius $r > a$.

$$\theta = \frac{a V}{r} \quad \text{erfc} \quad \frac{r-a}{2\sqrt{\alpha t}}$$

2. Heat flux at r = a

$$H = kV\left[\frac{1}{\sqrt{\pi \alpha t}} + \frac{1}{a}\right]$$

Nomenclature:

- θ = temperature above arbitrary datum plane, °F
- V = temperature at x=0 or r=a for constant temperature case, °F
- Q = constant heat flux, BTU/hr-ft^2
- H = heat flux (variable), BTU/hr-ft^2
- t = time, hrs
- k = thermal conductivity, BTU/hr-ft°F
- α = thermal diffusivity, ft^2/hr
- x = depth into semi-infinite solid, ft.
- a = radius of circular cylinder or sphere, ft.
- r = radius of concentric cylinder or sphere composed of solid at $r > a$, ft.

$B = \frac{\alpha t}{a^2}$, dimensionless

$$\operatorname{erfc} y = \frac{2}{\sqrt{\pi}} \int_y^\infty e^{-\beta^2}\, d\beta$$

$$\operatorname{ierfc} y = \frac{1}{\sqrt{\pi}}\, e^{-y^2} - y \operatorname{erfc} y$$

IV. REVISIONS OF ENGINEERING MANUAL

Part XVI - Chapter 3

Certain tentative revisions in the Engineering Manual, XVI Chapter 3, "Heating, Ventilating and Moisture Control", were made by B. A. Peavy of this Bureau and J. C. Letts of the O.C.E. The additions to the manual were mainly concerned with three conditions of occupancy for which the performance of an air conditioning system and related equipment must be designed. Some deletions were made in the manual because of their redundancy. Copies of the manual with these tentative revisions were presented to interested parties for their comment and review. Further work to be accomplished in writing of the manual will be concerned with work now being performed by this Bureau.

V. FUTURE EXPERIMENTATION - MT. WEATHER, VA.

1. Underground Test Chamber

Test Condition 14 - Test condition 3 repeat temperature drop with minimum heat supply and no ventilation.

Test Condition 15 - Apply refrigeration to chamber until room temperature reaches 40°F.

Test Condition 16 - Test condition 15 with use of ventilation air.

Test Condition 17 - Repeat test condition 1, constant heat input at about 75,000 BTU/hr until temperature of chamber is 75°F.

2. Underground Spray Pond Heat Exchanger

The spray pond was ready to operate before the finish of this period, but interference due to blasting by Bureau of Mines operations halted this test. Testing will begin May 11, with the test conditions for the various spray pond tests controlled as follows:

Test Condition 1 - A constant heat input rate to the spray water of approximately 60,000 BTU/hr will be used until the temperature of the pond water reaches 100°F.

Test Condition 2 - Maintain the temperature of the pond water at 100°F.

Test Conditions 3 and 4. Repeat test conditions 1 and 2 with use of stagnant pond instead of using sprays.

3. Tunnel Ventilation

Preparations are being made for determining the heat exchange between tunnel walls and an air stream. Tests will begin during the month of June.

VI. USE OF AN OCCUPIED UNDERGROUND SPACE FOR TEMPERATURE STUDIES.

Background:

Following a visit to an underground space near Fort Ritchie, Maryland, it was concluded that some of the data needed for correlation with present studies at Mount Weather, Va. could be obtained by making appropriate tests in the underground space there. Seasonal heating or cooling of the ventilation air by rock wall shafts will occur at this site and the amount of heat exchanged between ventilation air and the rock mass could be evaluated. Also studies of the heat transferred to the rock mass surrounding the occupied structure could be made.

Proposed Work to be Done:

Temperature sensing elements will be installed on the rock surface and at selected depths in the rock in the ventilation air shafts and inside and outside of the structure proper. Other instruments will be provided to measure air flow, heat flux to rock mass, and relative humidity. These instruments and their locations will be carefully selected so that measurements would provide data for computations of heat transfer rates. During this reporting period thermocouples were installed at six positions on the rock surface and at selected depths in the rock. The remaining work will be finished during the month of June.

TABLE 1

B	f (B)	B	f (B)
.001	.0321	0.050	.2302
.002	.0495	0.052	.2341
.004	.0694	0.054	.2382
.006	.0844	0.056	.2422
.008	.0971	0.058	.2461
.010	.1081	0.060	.2499
.012	.1176		
.014	.1269	0.065	.2591
.016	.1348	0.070	.2677
.018	.1430	0.075	.2765
.020	.1503	0.080	.2843
.022	.1572	0.085	.2921
.024	.1636	0.090	.2996
.026	.1700		
.028	.1760	0.10	.3138
.030	.1819	0.11	.3273
.032	.1859	0.12	.3401
.034	.1910	0.13	.3519
.036	.1963	0.14	.3633
.038	.2009	0.15	.3741
.040	.2074	0.16	.3847
.042	.2123	0.17	.3937
.044	.2210		
.048	.2258		

TABLE 2

Table of Functions Used in Heat Transfer Equations

y	erfc y	2 ierfc y	I (0, 1; y)
0.00	1.0000	1.1284	-
.01	.9887	1.1085	15.122
.02	.9774	1.0888	11.033
.03	.9662	1.0694	9.218
.04	.9549	1.0502	8.135
.05	0.9436	1.0312	7.359
.06	.9324	1.0124	6.846
.07	.9212	0.9939	6.421
.08	..9099	0.9756	6.076
.09	.8987	0.9575	5.790
0.10	.8875	0.9396	5.549
.11	.8764	0.9220	5.340
.12	.8633	0.9046	5.158
.13	.8542	0.8874	4.998
.14	.8431	0.8704	4.854
.15	.8320	0.0537	4.726
.16	.8210	0.8371	4.609
.17	.8101	0.8208	4.503
.18	.7991	.8047	4.405
.19	.7882	.7889	4.315
0.20	.7773	.7732	4.232
.21	.7665	.7578	4.155
.22	.7557	.7426	4.083
.23	.7450	.7275	4.016
.24	.7343	.7128	3.953
.25	.7237	.6982	3.894
.26	.7131	.6838	3.838
.27	.7026	.6697	3.785
.28	.6922	.6557	3.735
.29	.6818	.6420	3.688
.30	.6714	.6284	3.643
.31	.6611	.6151	3.600
.32	.6509	.6020	3.559
.33	.6408	.5891	3.520
.34	.6307	.5764	3.482
.35	.6206	.5639	3.446
.36	.6106	.5515	3.412
.37	.6008	.5394	3.379
.38	.5909	.5275	3.348
.39	.5813	.5158	3.317

TABLE 2 - continued

y	erfc y	2 ierfc y	I (0, 1; y)
.40	.5716	.5043	3.288
.41	.5620	.4929	3.259
.42	.5526	.4818	3.232
.43	.5431	.4708	3.206
.44	.5338	.4600	3.180
.45	.5245	.4495	3.156
.46	.5154	.4391	3.132
.47	.5063	.4289	3.109
.48	.4973	.4188	3.086
.49	.4884	.4090	3.065
.50	.4795	.3993	3.044
.51	.4708	.3898	3.023
.52	.4621	.3805	3.003
.53	.4535	.3713	2.984
.54	.4451	.3623	2.965
.55	.4367	.3535	2.947
.56	.4284	.3448	2.929
.57	.4202	.3364	2.912
.58	.4121	.3280	2.895
.59	.4041	.3199	2.878
.60	.3961	.3119	2.862
.61	.3883	.3040	2.847
.62	.3806	.2963	2.831
.63	.3729	.2888	2.816
.64	.3654	.2814	2.802
.65	.3580	.2742	2.787
.66	.3506	.2671	2.773
.67	.3434	.2602	2.760
.68	.3362	.2545	2.746
.69	.3292	.2467	2.733
0.70	.3222	.2402	2.720
.71	.3154	.2338	2.708
.72	.3086	.2276	2.695
.73	.3019	.2215	2.683
.74	.2953	.2155	2.671
.75	.2888	.2097	2.660
.76	.2825	.2040	2.648
.77	.2762	.1984	2.637
.78	.2699	.1929	2.626
.79	.2639	.1876	2.616

TABLE 2 - continued

y	erfc y	2 ierfc y	I (0, 1;y)
0.80	.2579	.1823	2.605
.81	.2519	.1772	2.595
.82	.2462	.1723	2.584
.83	.2405	.1674	2.574
.84	.2349	.1626	2.565
.85	.2293	.1580	2.555
.86	.2239	.1535	2.545
.87	.2186	.1490	2.536
.88	.2133	.1447	2.527
.89	.2082	.1405	2.518
0.90	.2031	.1364	2.509
0.91	.1981	.1324	2.500
0.92	.1932	.1285	2.492
0.93	.1884	.1247	2.483
0.94	.1837	.1209	2.475
0.95	.1791	.1173	2.467
0.96	.1746	.1138	2.459
0.97	.1701	.1103	2.451
0.98	.1658	.1070	2.443
0.99	.1615	.1037	2.435
1.00	.1573	.1005	2.427
1.02	.1492	.0944	
1.04	.1414	.0886	
1.06	.1339	.0831	
1.08	.1267	.0779	
1.10	.1197	.0729	2.357
1.12	.1132	.0683	
1.14	11069	.0639	
1.16	.1009	.0597	
1.18	.0952	.0558	
1.20	.0897	.0521	2.259
1.25	.0771	.0438	
1.30	.0660	.0366	2.240
1.35	.0562	.0305	
1.40	.0477	.0253	2.191
1.50			2.147
1.60			2.106
1.70			2.069
1.80			2.036
1.90			2.004

y	I (0, 1; y)
2.0	1.975
2.5	1.856
3.0	1.767
3.5	1.697
4.0	1.639
4.5	1.591
5.0	1.550
6.0	1.483
7.0	1.429
8.0	1.386
9.0	1.349
10.0	1.317
20.0	1.138
30.0	1.052
40.0	0.997
60.	0.928
80	0.884
100	0.853

VII. RESULTS OF AN INITIAL WARM-UP PERIOD - APRIL 23 TO MAY 15, 1952.

Object:

The object of this test was to determine the time needed to bring the temperature of an underground chamber up to a temperature within the human occupancy comfort range by the means of a constant heat input rate, and also to determine empirically the equations of heat transfer to the mass bounding the chamber and the relation of these empirical equations to the theoretical approaches listed in Part III of this report.

Description of Underground Chamber and Equipment:

The underground chamber, the dimensions of which are 100'x35'x10' high, is contained in and adjacent to an experimental mine operated by the Bureau of Mines at Mount Weather, Virginia. The chamber is approximately 215 feet below the surface of the ground and 1200 feet from the surface in a horizontal direction. The rock mass bounding the chamber consists mainly of greenstone with a scattering of epidote and quartz, and traces of various other minerals. Petrographically, the greenstone is a metamorphic basalt partially colored green by the presence of chlorite.

Measurements of the surface area of the walls, floor and ceiling were made and showed that the projected surface area was approximately 10,000 square feet. Physical determinations of greenstone rock from the excavation were made at this Bureau and the results were:

Density, ρ	= 186 lbs/ft^3
Specific heat, c	= 0.2 BTU/lb. °F
Thermal Conductivity (cores), k	= 1.45 BTU/hr.ft.°F
Thermal diffusivity, α	= 0.039 ft^2/hr

The apparent porosity of greenstone samples tested by the Bureau of Mines was 0.50 percent.

As shown in Figures 11 and 12, the walls and ceiling are painted white and the floor paved with concrete. Figure 1 shows plan and elevation views of the chamber and arrangement of mechanical equipment and air distribution ductwork. Air was forced by the circulating fan (1) into the ductwork past electric strip heaters (5) to the diffusers (6) and the air from the chamber was returned to the circulating fan through the air return filters (7) and the plenum chamber.

Fifteen twelve-foot long thermocouple poles were placed at selected positions (Figure 2) in the rock. Thermocouples had been previously attached to these poles at intervals--one half foot intervals up to six feet and one foot intervals from six to twelve feet. Room air temperatures were measured at twenty plan positions and at each plan position at heights of 2, 30, 60, and 90 inches from the floor. The thermocouples were copper-constantan and temperatures were measured by them in conjunction with an indicating potentiometer located in an instrument shed built within the chamber.

Test Procedure:

With the initial temperature in the rock up to 12 feet in depth practically uniform at 53.5°F, a constant heat input of 17.8 kilowatt or 60,800 BTU/hr was supplied to the test chamber. At regular intervals during the test, temperatures of the rock surface, rock depth room air, wet and dry bulb were recorded as well as the electric energy input as measured by kilowatt-hour meters.

The test was arbitrarily terminated at the time when the average plane rock surface temperature reached 70°F. This time was 522 hours or 21.75 days.

Results:

Figure 3 is a plot of the average rock temperature (computed from the average of the temperatures on the fifteen poles) against depth in the rock, with time from the atart of the test as a parameter. Figures 4-8 show temperature distributions at various crossections in the rock.

Figure 9 is a log-log plot of time against average temperature rise above the initial rock temperature of 53.5°F with depth in the rock as a parameter, and also the average temperature rise of the room air with time. The average room air temperature was approximately 6°F above that of the rock surface temperature throughout the test and like the rock surface temperature rise plots as a substantially straight line on log-log paper. The heat transfer from the room air to the rock surface appears to obey Newtons Law:

$$Q = h \Delta T \qquad (1)$$

where the coefficient 'h' in this case was approximately 1.0 BTU/hr ft^2°F.

An empirical equation of average rock surface temperature rise with time has been computed from the data by the method of averages, namely:

$$\theta = 0.69t^{1/2} \tag{2}$$

Referring to Part III of this report, equations relating temperatûre rise at the rock surface, with the elapsed time of heating for a constant heat flux into a mass approaching infinite thickness, are for the plane surface, and the cylindrical case:

$$\theta = \frac{2\,Q}{k} \sqrt{\frac{\alpha t}{\pi}} \tag{3}$$

$$\theta = \frac{2\,Q}{k} \left[\sqrt{\frac{\alpha t}{\pi}} - \frac{\alpha t}{4a} + \frac{(\alpha t)^{3/2}}{4a^2 (\pi)^{1/2}} - \frac{3\,(\alpha t)^2}{32\,a^3} \right] \tag{4}$$

respectivěly (for nomenclature refer to Part III of this report), where the first term in the cylindrical case is the same as that for the plane surface.

Using the experimental surface temperature data, the constants determined at the Bureau for thermal conductivity and diffusivity, and assuming the equivalent cylindrical radius of the chamber was the average of the radii computed from a) the perimeter and b) the crosssectional area, the heat flux was determined to be 4.49 and 4.90 BTU/hr.ft^2 for the plane surface and cylindrical case, respectively. Using these values for determining temperatures at a depth in the rock the calculated values are compared with the actual experimental values in Figure 10.

Following is a table showing the heat input to the chamber from readings of watt hour meters compared with the heat stored in the rock computed from the mean temperature rise at various times from the start of the test. Also the depth of perceptible heat penetration is noted.

Time, Hrs.	Electric Heat Input, BTU	Heat in Rock BTU	Heat Penetration ft
49	3,010,300	2,900,000	5.5
100	5,960,000	6,700,000	7.8
170	10,256,000	11,450,000	9.0
290	17,928,000	19,020,000	11.2

Discussion and Conclusion:

1. For the duration of this test, the temperature rise at the rock surface was proportional to the square root of the time (Equation 2). This is substantiated by theory (Equations 3 and 4) where the second, third and fourth terms of equation 4 are small in comparison to the first term for small values of time.

2. The coefficient of heat transfer between the air and rock was approximately 1.0 BTU/hr ft^2(deg F). This value is approximately what would be expected for this case wherein heat transfer was by natural convection from nearly still air, with no radiation because all surfaces were nearly at the same temperature.

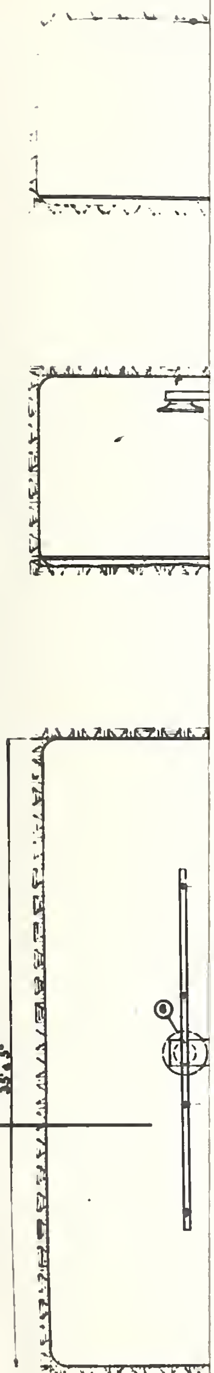
3. The heat balance showed that the heat stored in the rock as computed from the observed temperatures of the rock agreed to within 5% of the measured electrical heat input.

4. Figure 10 shows that the use of the equation for heat transfer to a medium surrounding a cylinder gave better agreement with the experimental data than the equation for heat transfer from a plane surface to a semi-infinite medium. (Refer to Part III of this report, sections I,A,1 and I,B,1.) For the cylindrical case, the agreement between the experimental and computed values was improved with increase in depth from the exposed rock surface.

5. Figures 4 through 8 show isotherms in the rock at various crossections. The isotherms tend to approach elliptical shape, especially in the smaller crossections.

6. The constant rate of energy input to the chamber, as measured electrically, was 60,800 BTU/hr, which gives an average heat flux of 6.08 BTU/hr ft^2 for the measured 104 square feet of projected surface area. The heat flux computed from the heat transfer equations on the basis of surface temperatures and measured rock properties was 4.49 and 4.90 BTU/hrft^2 for the plane surface and cylindrical equations, respectively. The discrepancy between measured and calculated heat flux is believed to be due to the fact that heat flow from the chamber surfaces took place in three dimensions (diverging as do the radii of a sphere), whereas the equations apply strictly to one-dimensional flow (plane equation) or two dimensional flow (cylinder

equation). The difference between the measured average input flux and that calculated from the equations is considered due to the extra bulk of rock beyond edges and corners, which caused greater flux at nearby surface areas than at other areas of the chamber, thus causing the average flux to be greater than the value at other areas where the stated equations apply more correctly. For the chamber investigated, the flux which appears appropriate for use in the cited equations was about 80% of the measured input flux.



equations. The difference between the measured average level flow and that calculated from the equations is considered due to the effect [illegible] [illegible] edges and corners, with [illegible] [illegible] [illegible] other areas of the chamber, [illegible] the average flow [illegible] than the value [illegible] [illegible] [illegible] [illegible] [illegible] [illegible] [illegible] the [illegible] was about 70% of the [illegible] flow.

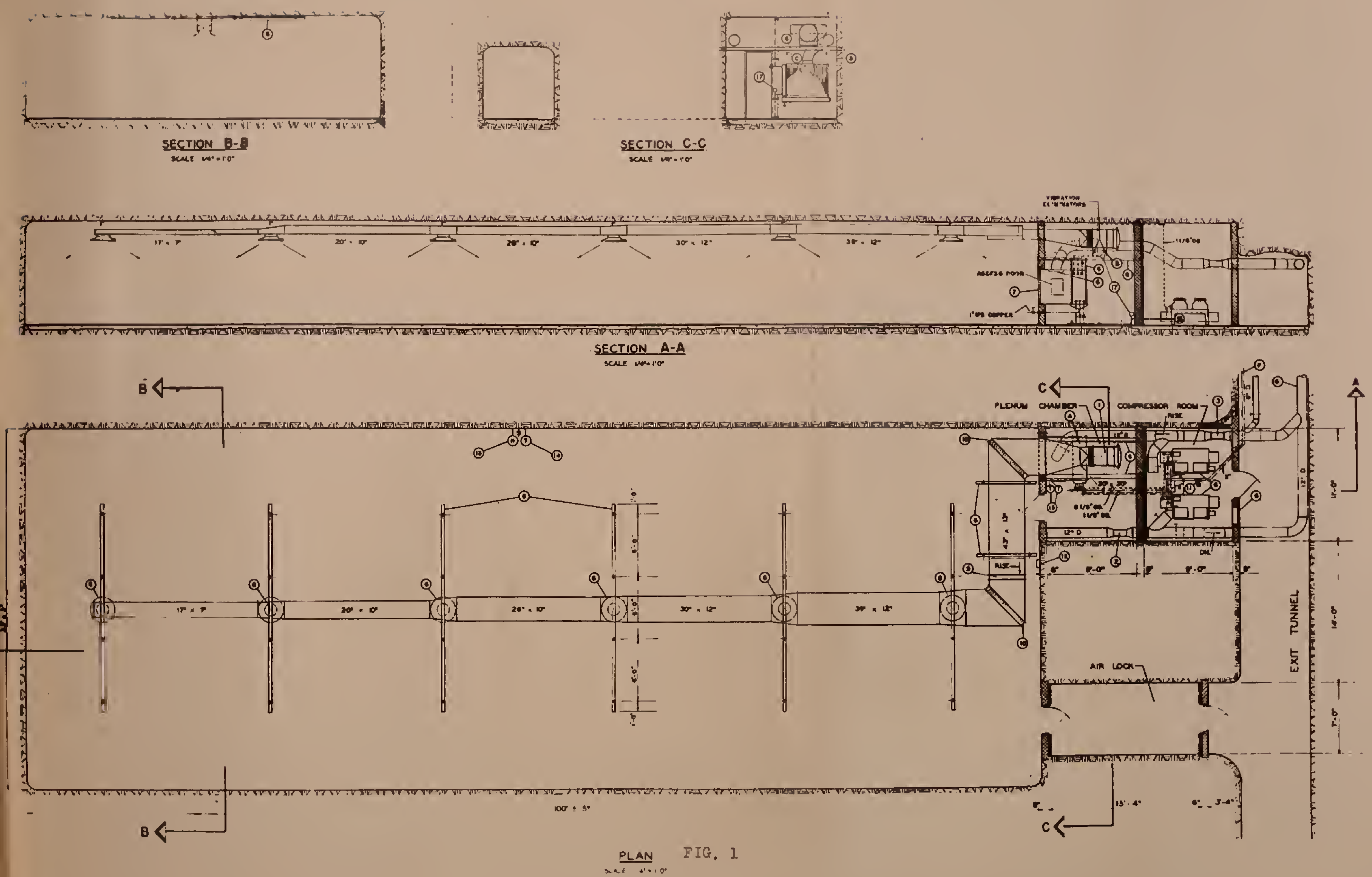


FIG. 1

CONSTRUCTION NOTES

A ALL SLEEVE & LINTEL OPENINGS PROVIDED BY OTHERS
B FAN Nº1 SUPPORT - 2-7.2 LB ANGLES 4" - 3" - 7/16" 4" LEG DOWN
C COOLING COIL SUPPORT - 2 ⊔'s 5" x 1-3/4" 6.7 LB
D COOLING COIL SUPPORT - 4 - L's 1 1/2" x 1 1/2" x 3/16" 1.8 LB
E DUCTWORK INSTALLATION TO TERMINATE AT THIS POINT INSTALLATION TO OUTSIDE BY OTHERS
F CONDENSER WATER PIPING TO TERMINATE AT THIS POINT INSTALLATION TO COOLING TOWER BY OTHERS
G DUCTWORK & CONDUIT SUPPORT - 2 - L's 3" x 3" x 1/4", 20 FT LONG INSTALLATION BY OTHERS
H DUCTWORK SUPPORT - 2 - L's 3" x 3" x 1/4", 5 FT LONG INSTALLATION BY OTHERS

EQUIPMENT LIST

ITEM Nº	DESCRIPTION	QUANTITY
1	FAN AXIAL - 6000 CFM 2" TP 440 V 3φ 60~	1
2	FAN AXIAL - 800 CFM 1.5" TP 440 V 3φ 60~	1
3	FAN AXIAL - 800 CFM 1.5" TP 440 V 3φ 60~	1
4	COOLING COIL 10φ SEE DETAIL SHEET M-3	1
5	STRIP HEATER 440 V 3φ 60~ SEE DETAIL SHEET M-3	1
6	DIFFUSERS, 1000 CFM EACH	6
7	AIR FILTERS 5500 CFM (20" x 25")	4
8	AIR FILTERS 500 CFM (20" x 20")	2
9	10 HP SELF CONTAINED WATER CHILLER	2
10	VANED TURNS	2
11	COOLING TOWER PUMP	1
12	STEP CONTROLLER	1
13	HUMIDISTAT	1
14	MODULATING THERMOSTAT (15-90°F)	1
15	MODULATING THERMOSTAT DUCT TYPE (15-85°F)	2
16	CHILLED WATER CONTROL (0-70°F)	1
17	THERMOMETERS (SEE DRAWING M-2)	6

REVISION	DATE	APP'D	DESCRIPTION	BY

MECHANICAL PLAN

NATIONAL BUREAU OF STANDARDS
WASHINGTON, D.C.

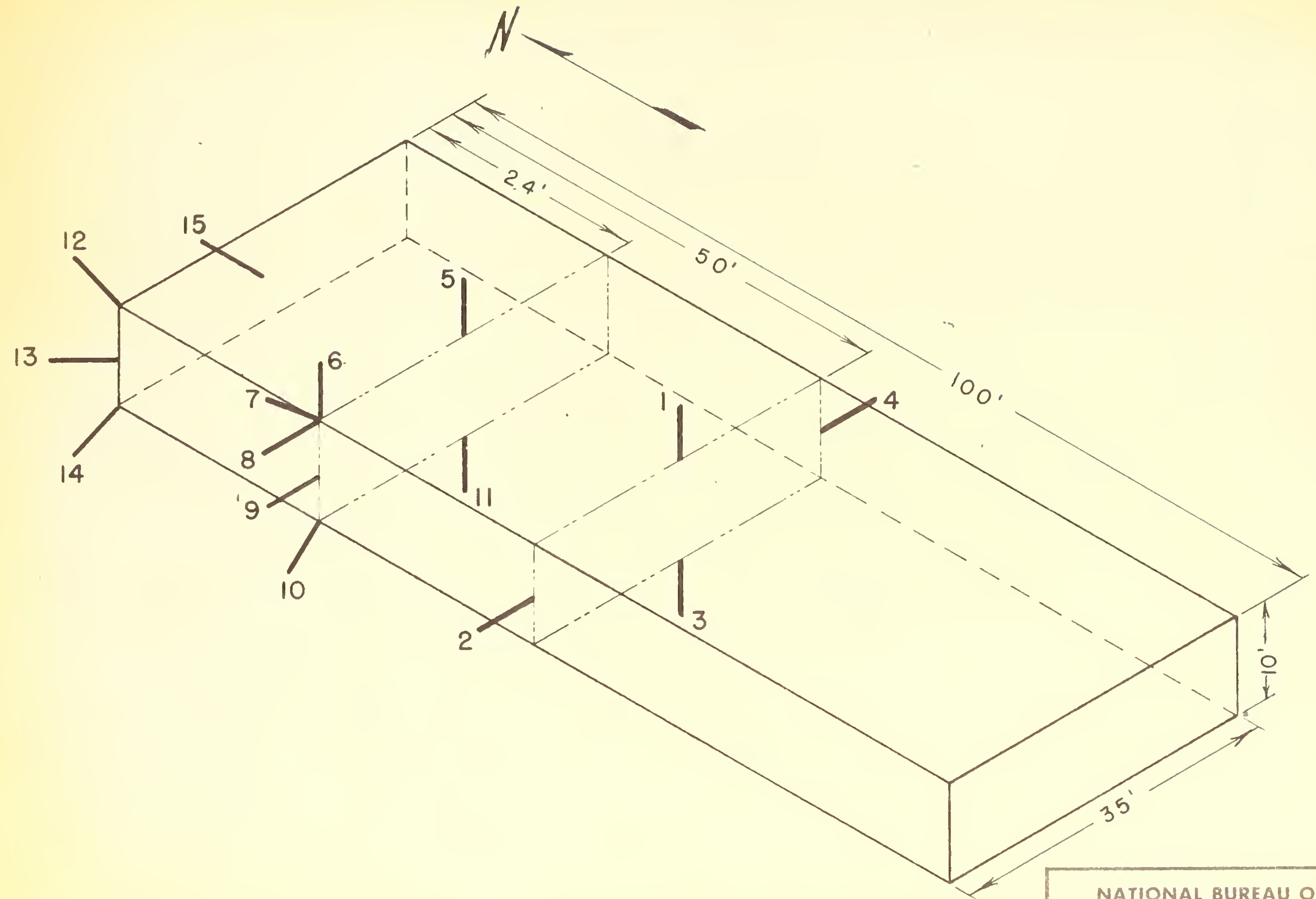
DESIGNED BY T.H.U.
DRAWN BY G.B.
TRACED BY G.B.
CHECKED BY T.H.U.
APPROVED

UNDERGROUND STRUCTURE PILOT PROJECT
FOR
THE OFFICE OF THE CHIEF OF ENGINEERS
U.S. ARMY

APPROVED
DATE MAY 5 1951

THOMAS H. URDAHL
CONSULTING ENGINEER
1734 JACKSON PLACE N.W.
WASHINGTON, D.C.

5 55 M 1

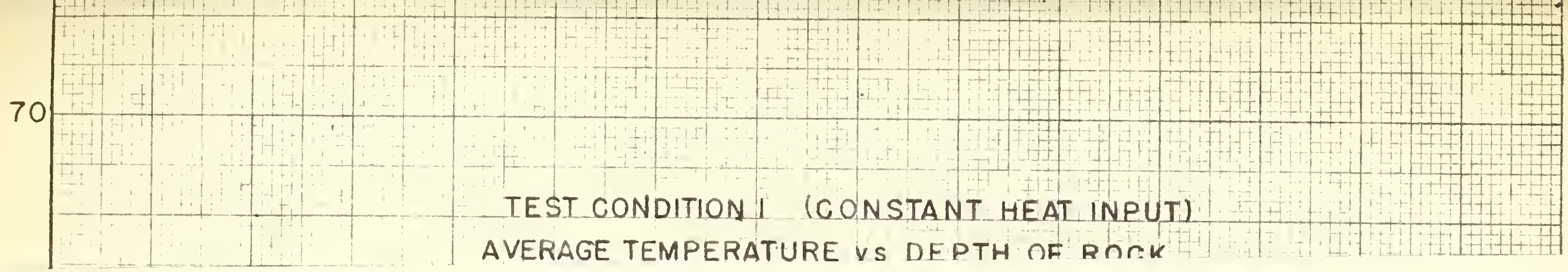


NATIONAL BUREAU OF STANDARDS
WASHINGTON 25, D. C.

DRAFTSMAN W. C. G. | DATE | SCALE
DIV | SEC

UNDERGROUND LABORATORY

FIGURE 2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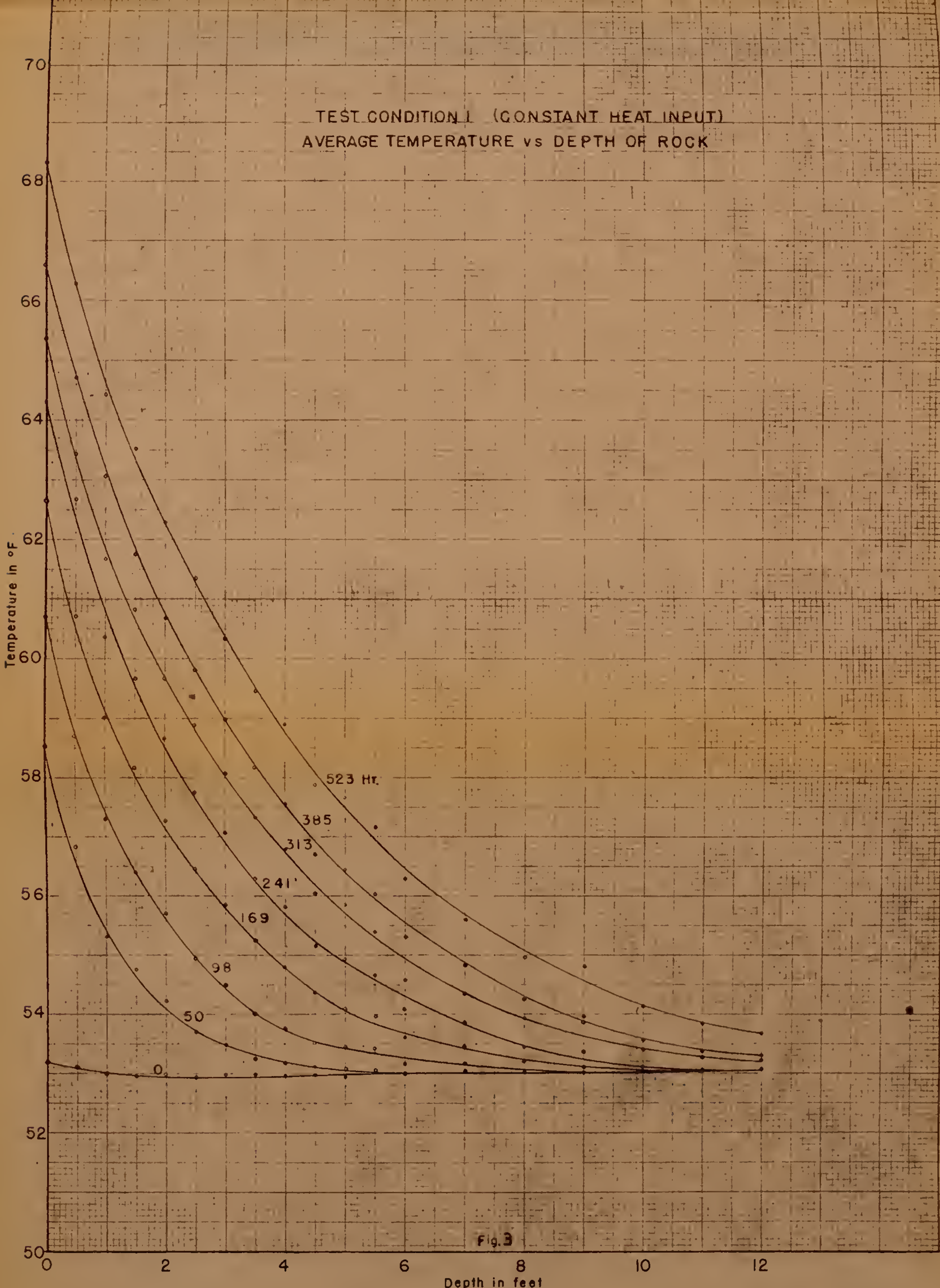
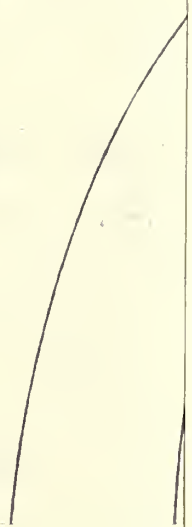
TEST CONDITION I (CONSTANT HEAT INPUT)
AVERAGE TEMPERATURE vs DEPTH OF ROCK

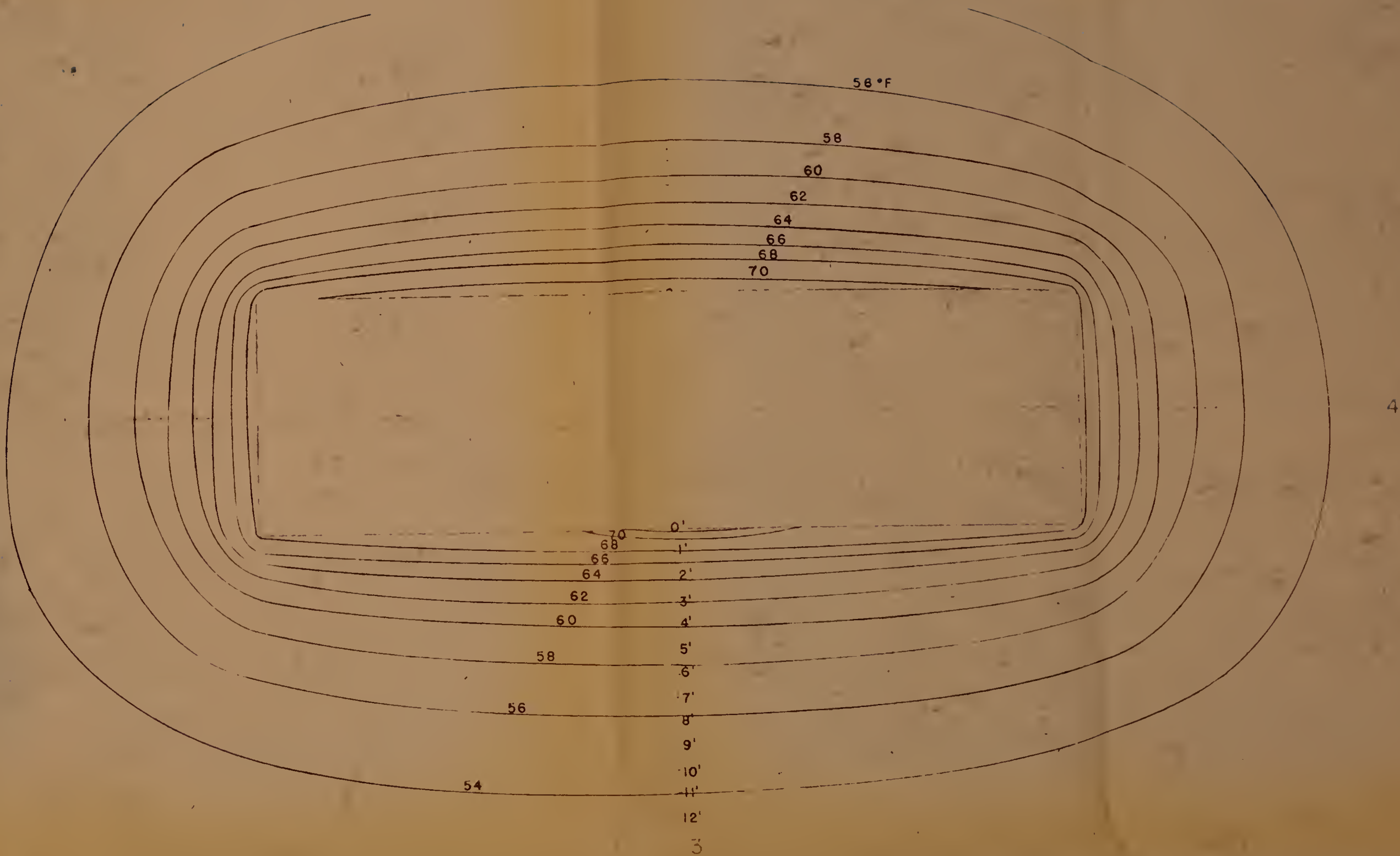


TEST CONDITION I (CONSTANT HEAT INPUT)
AVERAGE TEMPERATURE vs DEPTH OF ROCK
Temperature in °F
70
68
66
64
62
60
58
56
54
52
50
523 Hr.
385
313
241
169
98
50
0
0
2
4
6
8
10
12
Depth in feet

Fig. 3

2





SECTION A-A

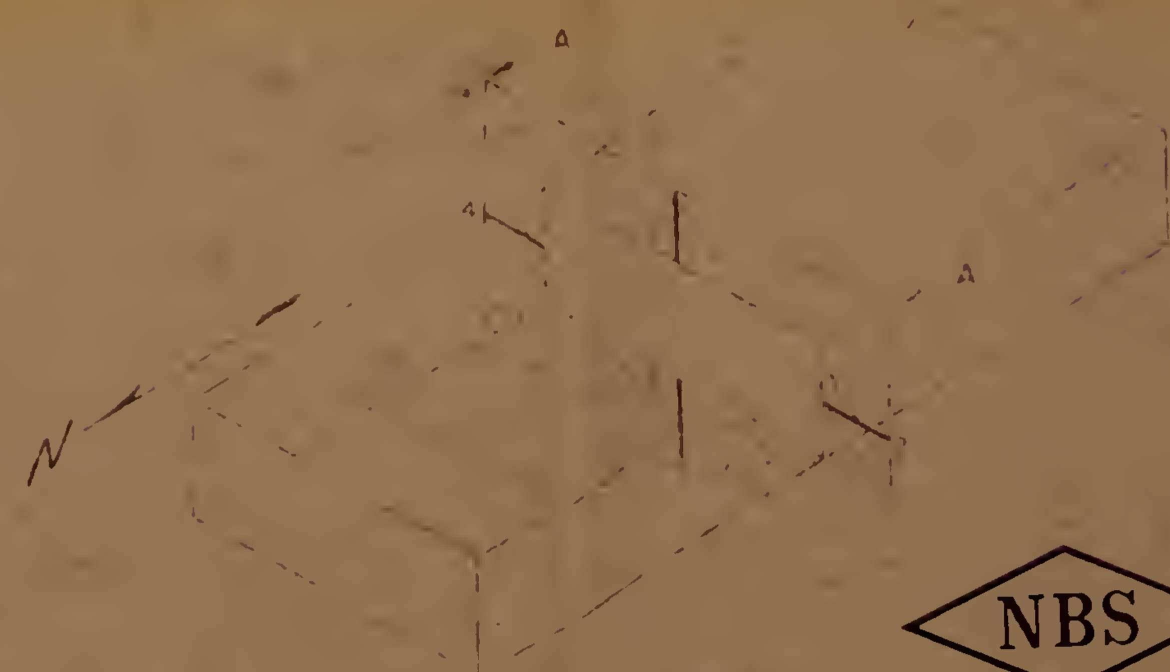
SCALE 1/4" = 1'-0"

TEST CONDITION No I - HEATING UP PHASE

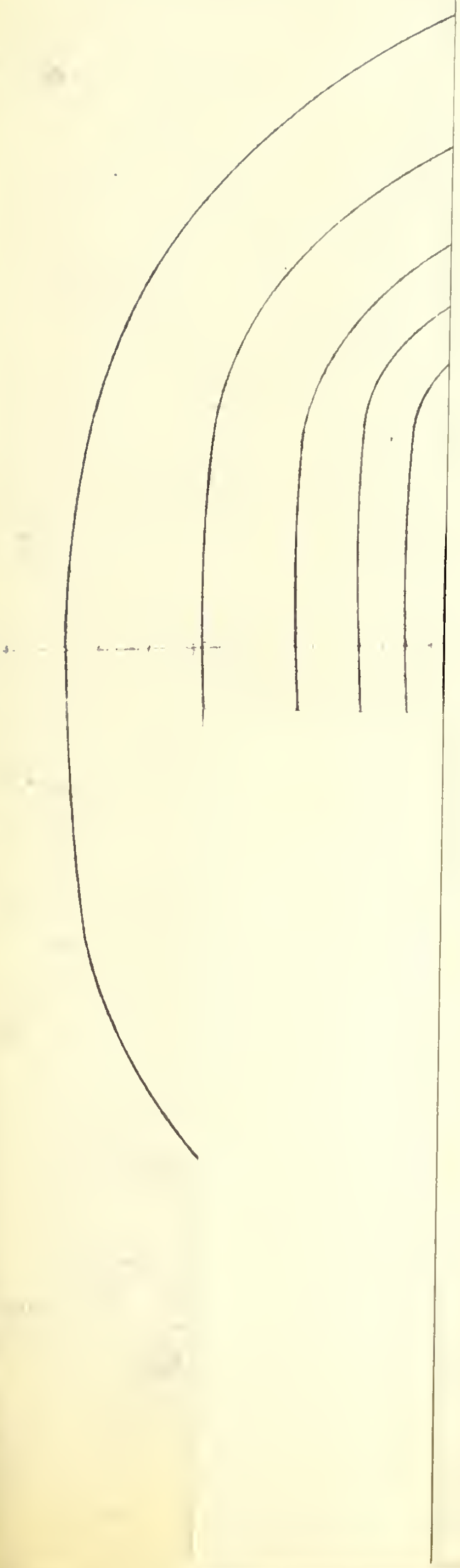
TEMPERATURE DISTRIBUTION IN ROCK

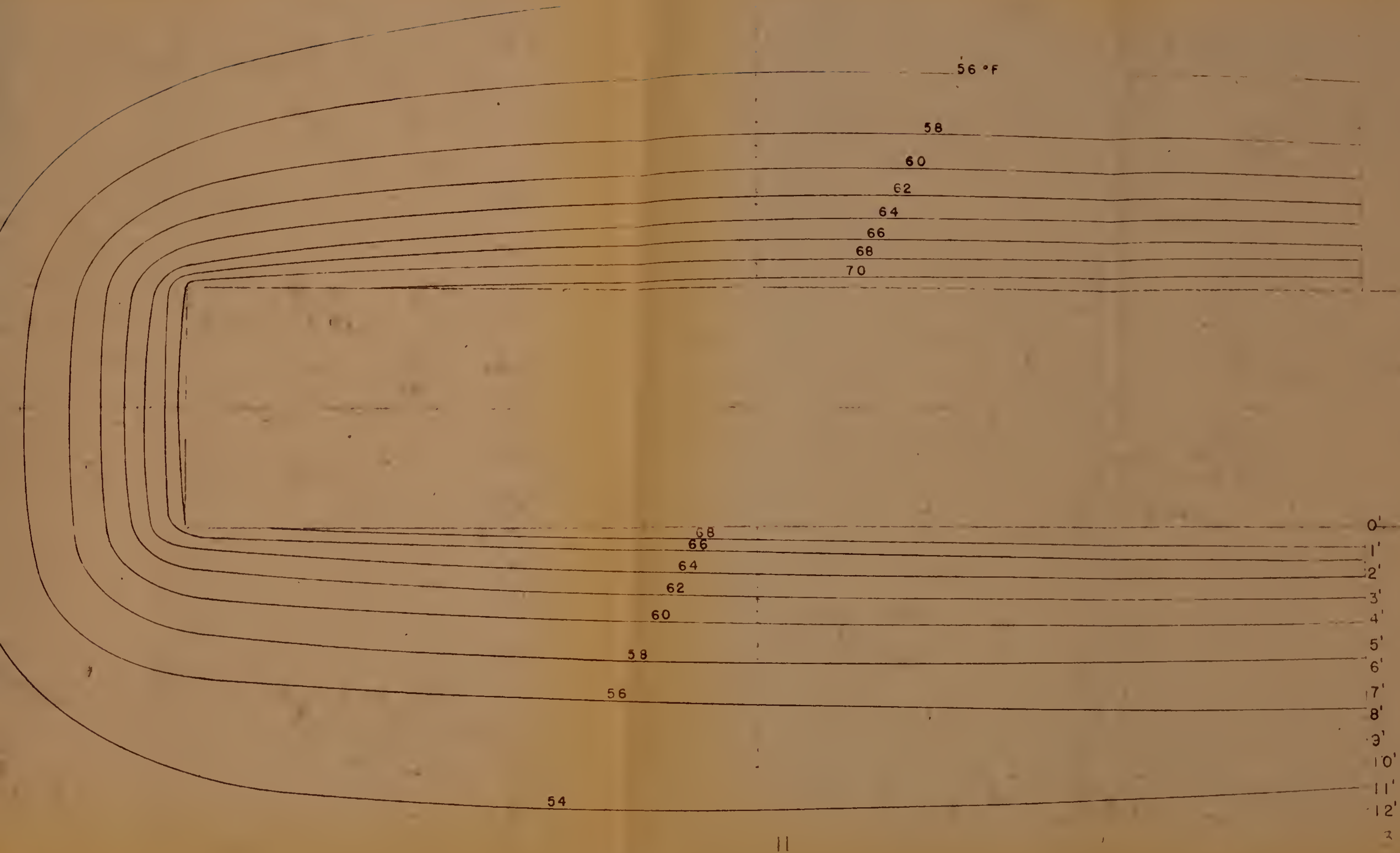
AFTER 521 HOURS

FIGURE 4



NBS





SECTION B-B

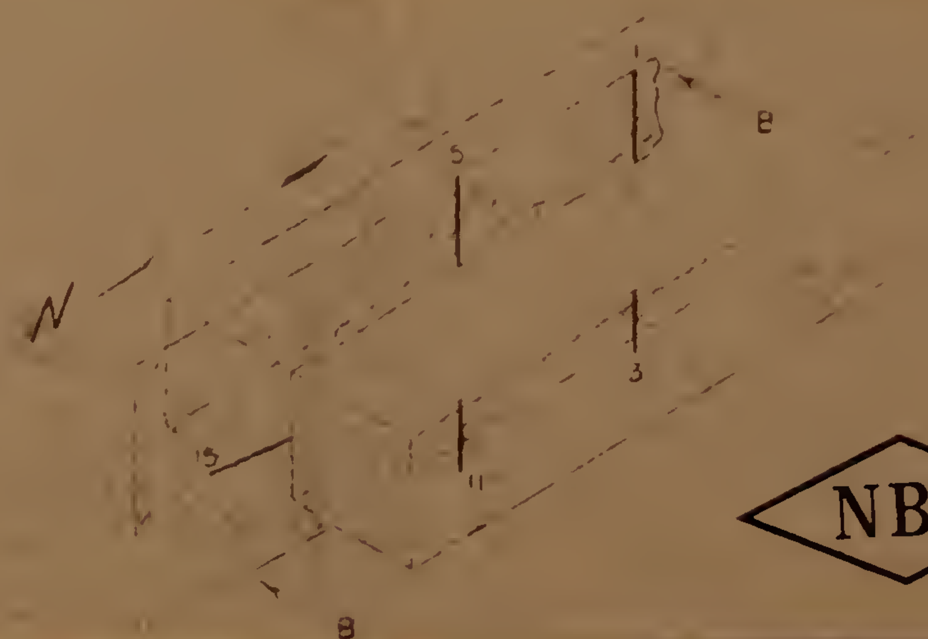
SCALE: $\frac{1}{4}$" = 1'-0"

TEST CONDITION No. 1 – HEATING UP PHASE

TEMPERATURE DISTRIBUTION IN ROCK

AFTER 521 HOURS

FIGURE 5



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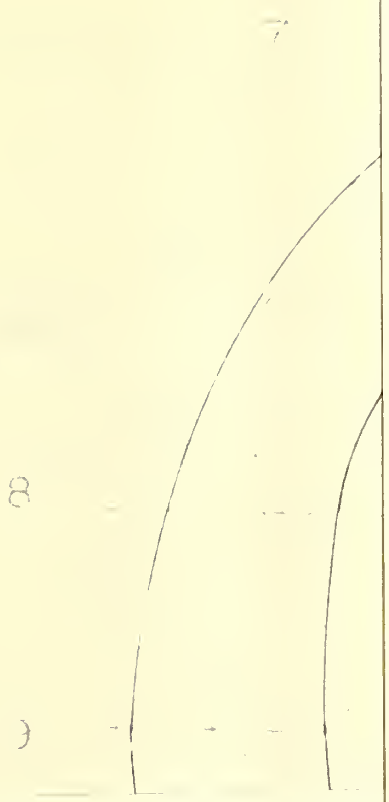
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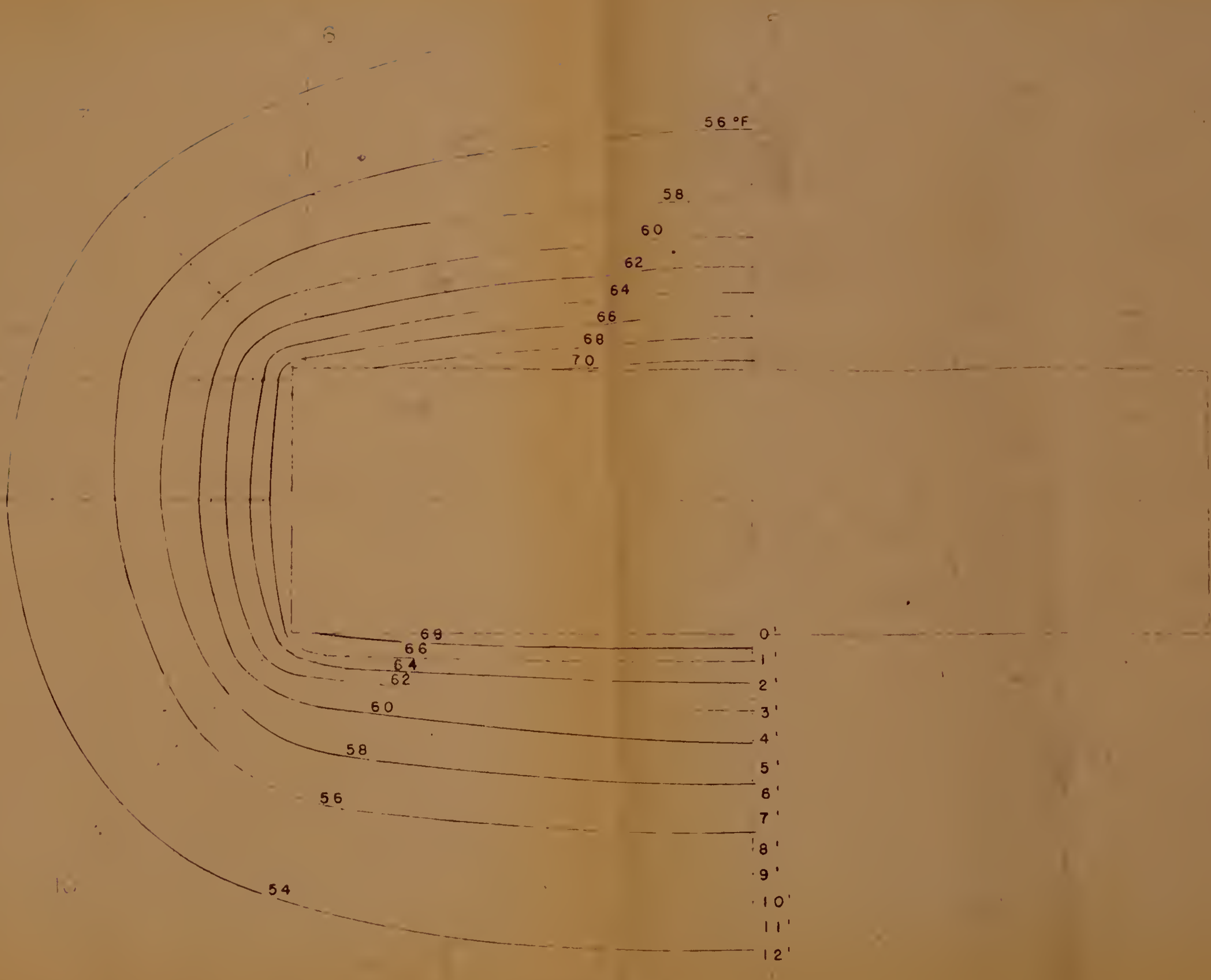


SECTION C-C

SCALE 1/4" = 1'-0"

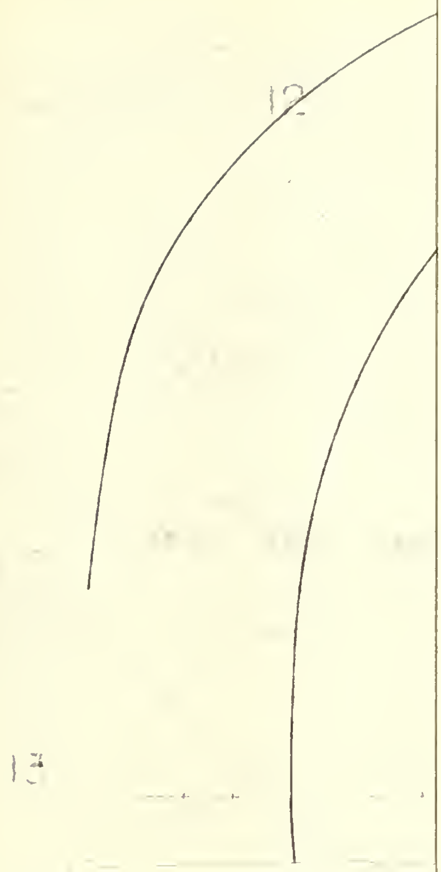
TEST CONDITION No. I – HEATING UP PHASE

TEMPERATURE DISTRIBUTION IN ROCK

AFTER 521 HOURS

FIGURE 6

NBS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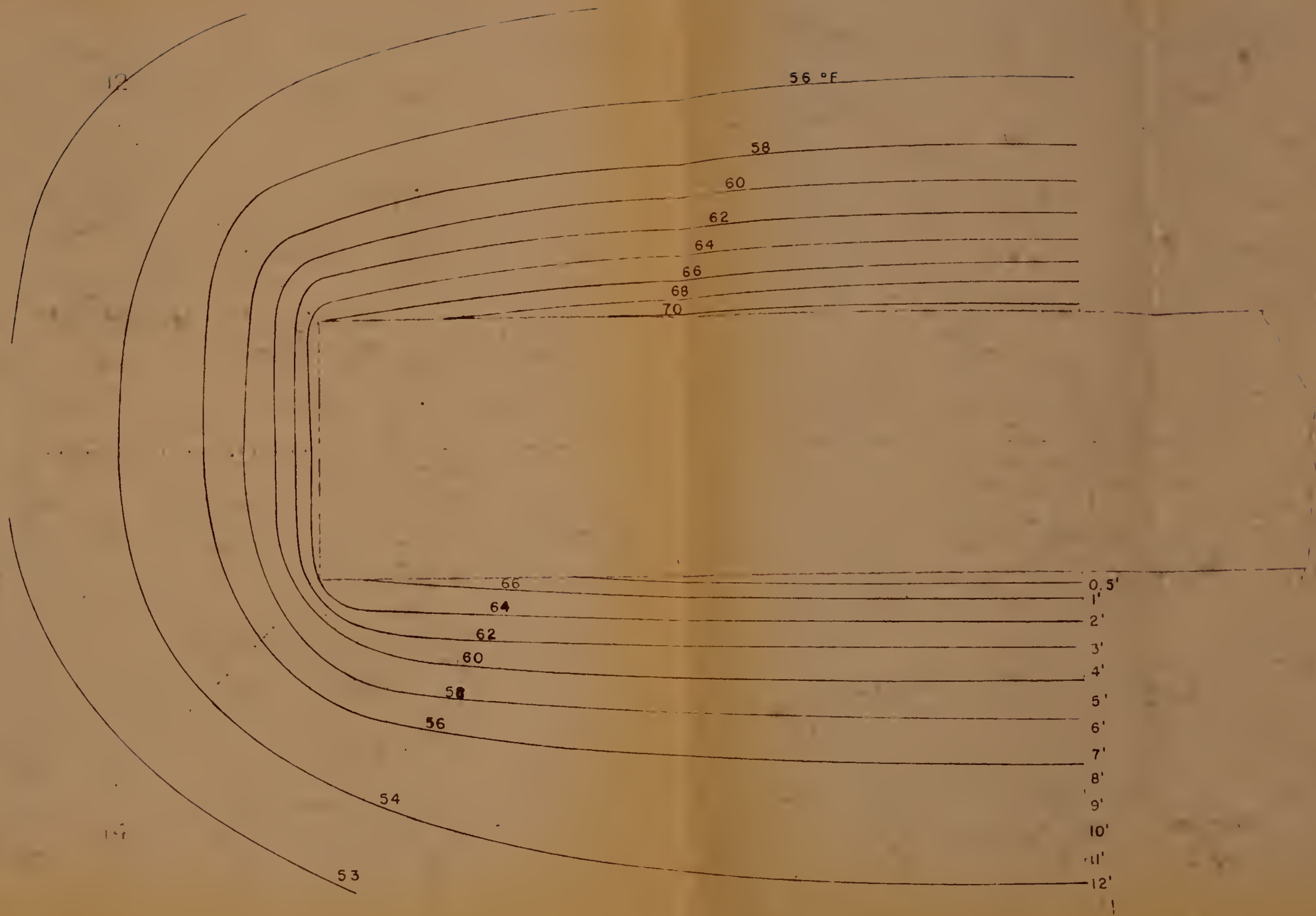
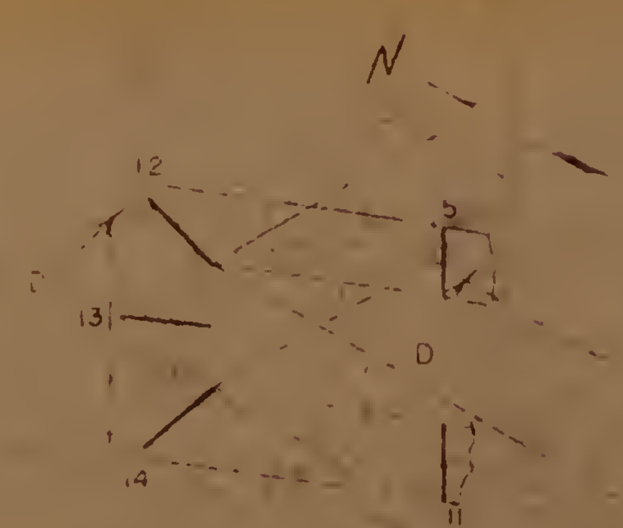
13



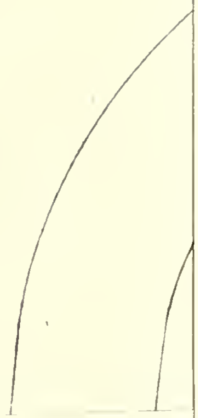


SECTION D-D

SCALE $\frac{1}{4}$" = 1'-0"

TEST CONDITION No. 1 – HEATING UP PHASE
TEMPERATURE DISTRIBUTION IN ROCK
AFTER 521 HOURS

FIGURE 7



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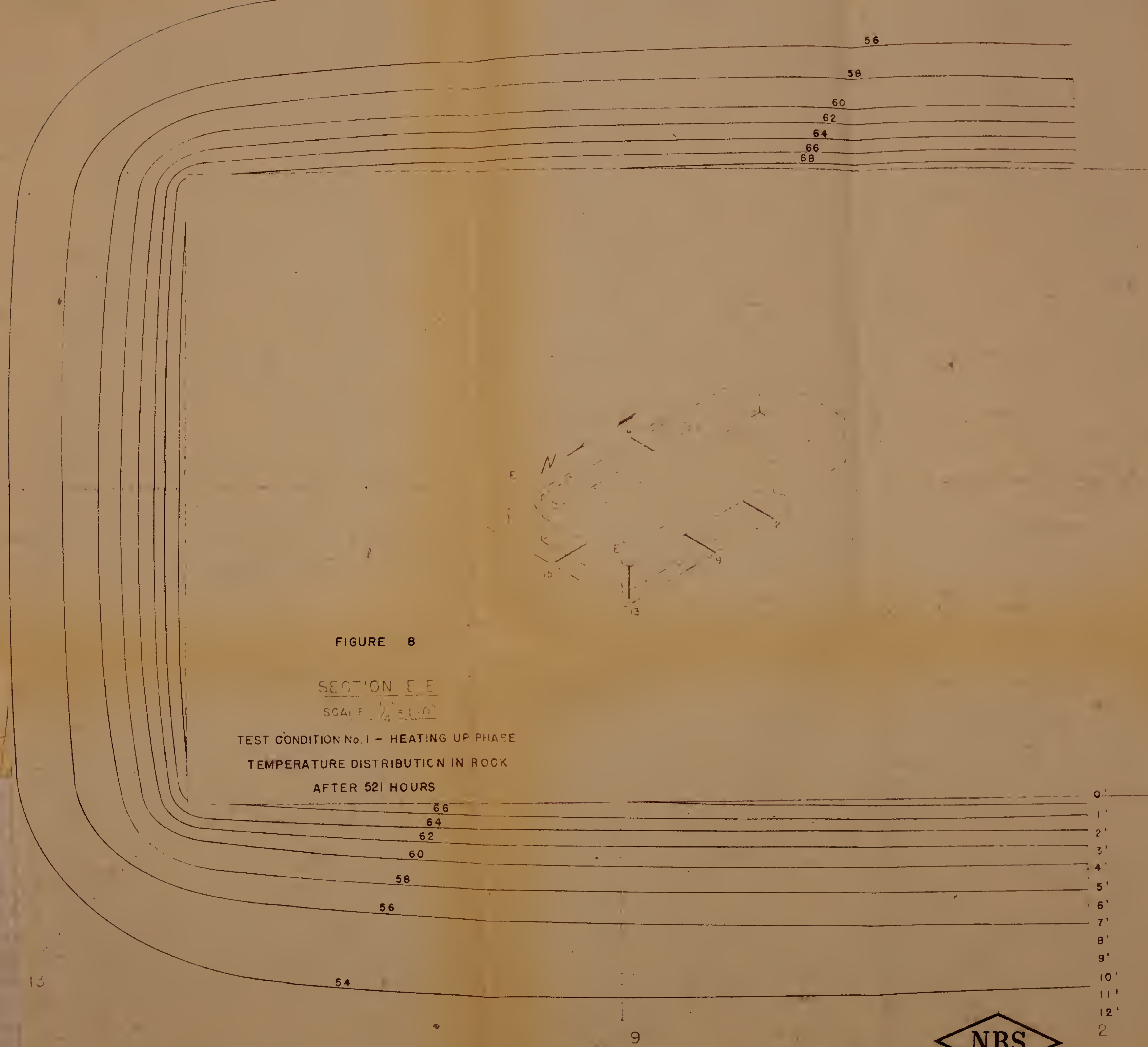

FIGURE 8

SECTION E-E

SCALE 1/4" = 1'-0"

TEST CONDITION No. 1 – HEATING UP PHASE

TEMPERATURE DISTRIBUTION IN ROCK

AFTER 521 HOURS

FIGURE 10

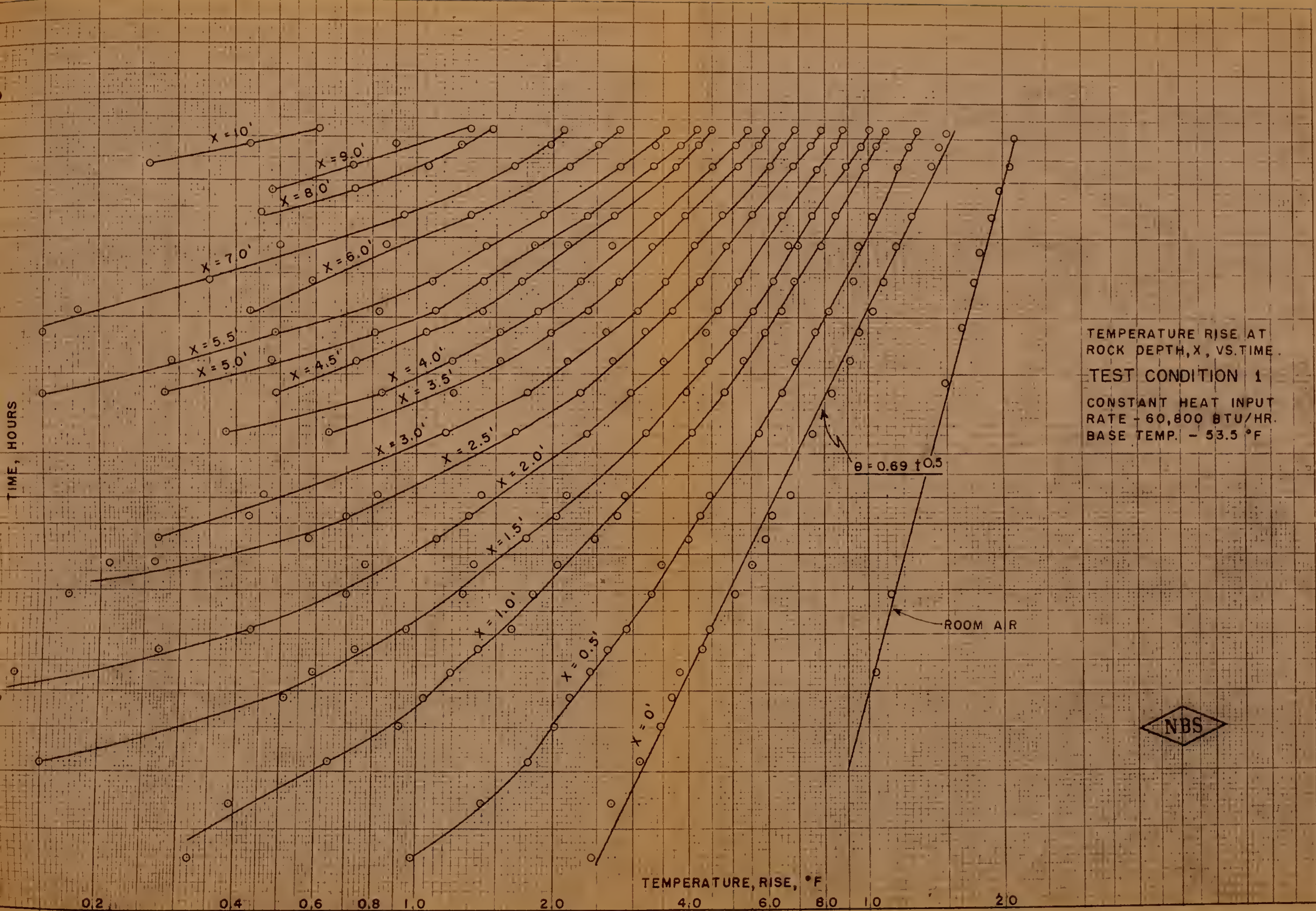
TEMPERATURE RISE AT ROCK DEPTH, X, VS. TIME.
TEST CONDITION 1
CONSTANT HEAT INPUT RATE - 60,800 BTU/HR.
BASE TEMP. - 53.5 °F
θ = 0.69 t^0.5
ROOM AIR
X = 10'
X = 9.0'
X = 8.0'
X = 7.0'
X = 6.0'
X = 5.5'
X = 5.0'
X = 4.5'
X = 4.0'
X = 3.5'
X = 3.0'
X = 2.5'
X = 2.0'
X = 1.5'
X = 1.0'
X = 0.5'
X = 0'
TIME, HOURS
TEMPERATURE, RISE, °F
0.2
0.4
0.6
0.8
1.0
2.0
4.0
6.0
8.0
10
20
NBS

FIG. 9

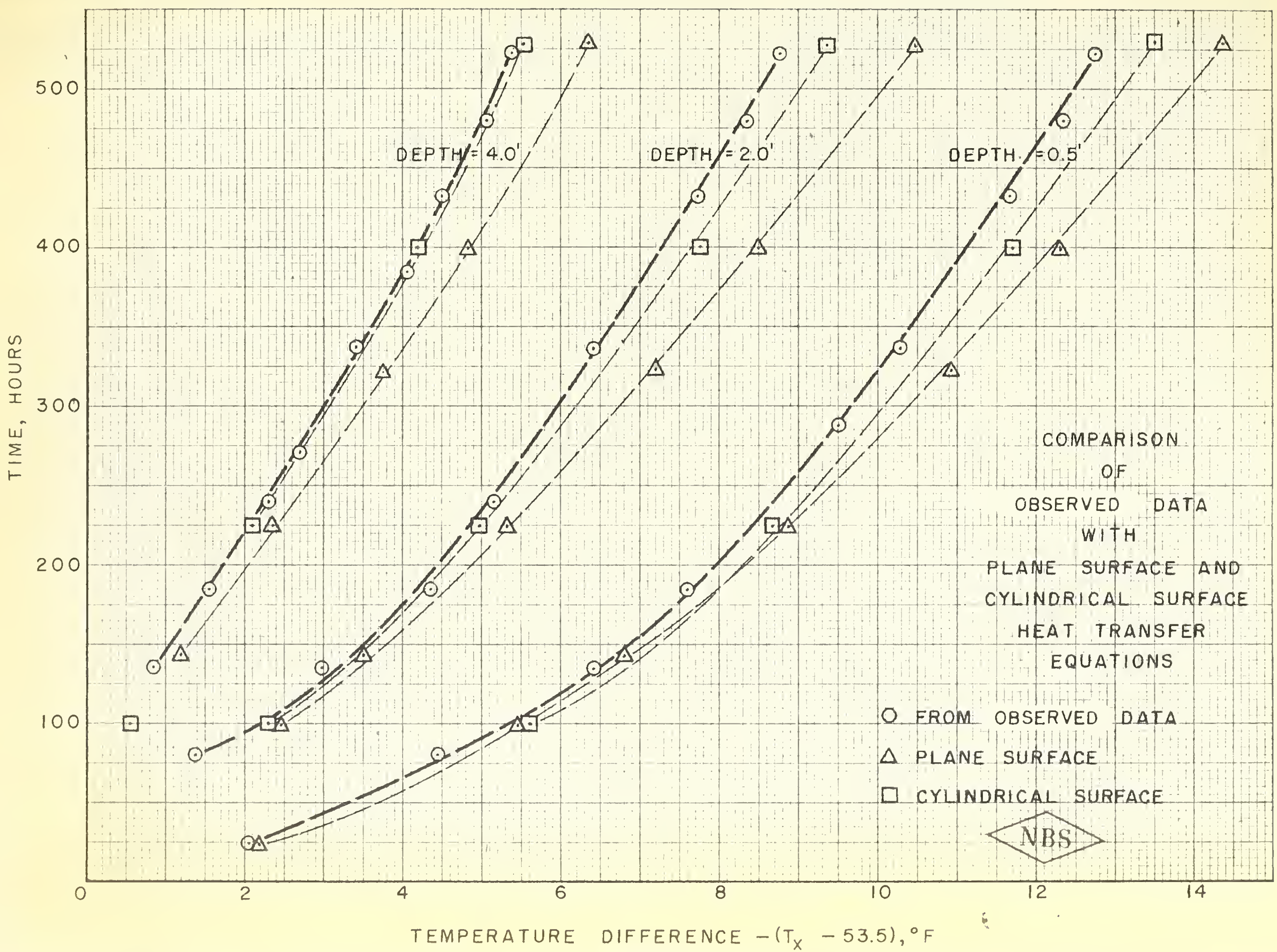
TIME, HOURS
500
400
300
200
100
0
2
4
6
8
10
12
14
DEPTH = 4.0'
DEPTH = 2.0'
DEPTH = 0.5'
COMPARISON
OF
OBSERVED DATA
WITH
PLANE SURFACE AND
CYLINDRICAL SURFACE
HEAT TRANSFER
EQUATIONS
FROM OBSERVED DATA
PLANE SURFACE
CYLINDRICAL SURFACE
NBS
TEMPERATURE DIFFERENCE $-(T_X - 53.5)$, °F

FIGURE 10

Figure 11. Underground Chamber - Entrance door on right, door to plenum chamber on left.

Figure 12. Underground Chamber - Rear